EFFECTS OF SLOT POSITION AND COOLANT INCIDENCE ANGLE ON THE COOLING PERFORMANCE OF VANE ENDWALL

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ABSTRACT
High pressure guide vane and the accompanying endwall are extensively cooled through discrete holes and leakage flow from the combustor-turbine interface gap. For making better use of the leakage flow, this paper described a short slot, where the leakage flow leaves the cavity. The numerical investigation of adiabatic effectiveness was compared with the short slot and long slot. This paper reports the effect of the slot axial position and coolant incidence angle on the cooling performance of endwall in a high pressure guide vane. According to previous investigation, leading edge region and pressure side-endwall junction are two of the most tough areas to supplement cooling air. With the slot flow injection, because of the effective suppression of the strength of secondary flow, the leading edge region and pressure side-endwall junction can be effectively cooled especially moving the short slot closer to the vane leading edge.

INTRODUCTION
The current demands for increasingly higher performance of modern gas turbine is directly achieved by increasing the turbine inlet temperature. With the increasing temperature, particularly the first stage turbine vane endures high thermal load. One critical region is vane endwall. Most combustor-turbine junctions have slots through which coolant leaks into the main gas path. These junctions, depending on the design and operating conditions, typically consist of either a backward-facing slot, a forward-facing slot or a flush slot. At the endwall the interaction of coolant air and secondary flow plays a crucial role on film cooling effectiveness. Secondary flow studied by Langston(Langston et al., 1977), Sharma and Butler(Sharma and Butler, 1987), Goldstein and Spores(Goldstein and Spores, 1988), and others shows to some extent similar structure. The incoming boundary layer on the endwall rolls up into horseshoe vortex at the leading edge. Then the horseshoe vortex splits into suction and pressure side legs, where the pressure side leg develops into a passage vortex.

Several past studies have investigated the performance of leakage flows from upstream interface gaps on endwall effectiveness. Blair(Blair, 1974) first investigated the endwall film cooling and found a strong increase in heat transfer occurs due to the strong secondary flow fields at the suction side corner. Friedrichs(Friedrichs et al., 1998, Friedrichs et al., 1997, Friedrichs et al., 1996) gave a detailed endwall film cooling performance without upstream slot. The results indicated a strong interaction between secondary flow and coolant flow. Roy(Roy et al., 2000) performed the experiments and numerical simulations to determine the convective heat transfer distribution on the hub endwall of model turbine vane passage. Heat transfer measurement without secondary air injection revealed a high heat transfer region occurred over a small area around and upstream of the vane leading edge. Secondary air injection through slots upstream of the vane leading edge at a blowing ratio 1.3 greatly reduced the heat transfer in the regions near the vane leading edge and the vane pressure side in the passage. Axial location plays an important role on cooling performance of an upstream slot. Kost and Nicklas(Kost and Nicklas, 2001) and Nicklas(Nicklas and Nicklas, 2001) carried out the
thermodynamic and aerodynamic measurements in a linear turbine cascade with transonic flow field. Strong variations in heat transfer and film-cooling effectiveness due to the interaction of the coolant air and the secondary flow field were found. The result shows that the horseshoe vortex was strengthened by the ejection of coolant from the leakage gap at a mass flow ratio of 1.3%. This was attributed to the slot position, which located at the place of saddle point. Other study by Kost and Muller-Kost and Muller (2006) investigated the effect of moving the slot to 0.3Cax upstream of the vane leading edge. They found that moving the slot location properly can avoid intensifying the strength of the horseshoe vortex effectively. It could be proved that the saddle point of the upstream endwall boundary layer stagnation region is a sensitive region where the coolant ejection should be avoided. It also found that slot at 0.2Cax can provide better coverage even with less than half of the amount of coolant. Thrift and Thole (Thrift et al., 2011) also investigated the effect of slot axial position and orientation. Effectiveness results indicate a significant increase in area averaged effectiveness for the 45° slot relative to the 90° orientation. Moving the 90° slot closer to the vane leading edge was shown to improve local and area averaged effectiveness. Unlike the two further upstream locations, the size of the induced vortex was greatly reduced while the vortices intensity increased. A systematical experiment has been reported for different injection schemes upstream of the vane leading edge with a contoured endwall by Burd et al. (Burd and Simon, 2000, Burd et al., 2000) and Oke et al. (Oke et al., 2000). In those studies coolant was injected from a interrupted, flush slot that was inclined at 45° upstream of the vane. They also found that most of the coolant swept toward the suction side at the low slot flow rate. As they increased the percentage of the slot flow to 3.2% of the exit flow, the effectiveness of endwall cooling got better within passage. Knost and Thole (Knost and Thole, 2003) presented a computational result of the cooling effectiveness of two endwall film cooling hole patterns combined with a flush slot upstream of the vane. Comparison between predictions and experiments of a flush slot alone showed a reasonable agreement. The resulting endwall effectiveness from slot cooling alone showed a pattern that is quite non-uniform along the endwall with most of the coolant being swept toward the suction side of the vane. One could expect a burn-out near the vane-endwall junction would be happened if only having slot provides the coolant. They also concluded that the superposition method cannot be simply used to calculate the total effectiveness of the flush slot and endwall cooling holes. Moreover, using the superposition method resulted in an overprediction of cooling results thereby underpredicting component life.

Though many studies have investigated the detailed combustor-turbine slot cooling effectiveness and the interaction between coolant flow and second flow fields, more understanding of the different type of combustor-turbine slot is needed. Also, the characteristic of real engine condition is needed to reveal from the design perspective comparing with low speed laboratory environment. The short slot, which distributes discontinuously in circumferential direction, was investigated in this paper especially about the effect of coolant flow from this kind of slot on the formation of horseshoe vortex and passage endwall cooling effectiveness. The contribution of varying axial position of the slot to reduce the second flow is studied. Additionally, coolant incidence angle and blowing ratio are also discussed to extend the understanding of the endwall cooling performance.

**COMPUTATIONAL METHODOLOGY**

For this study all the simulations were performed using the ANSYS CFX commercial software package. ANSYS CFX is a three-dimensional compressible RANS finite-volume flow solver. Mentor’s SST k-ω turbulence model is selected with automatic wall function. The equations are discretized using a finite volume method with the convective variables being resolved in a second-order upwind scheme. The RANS, energy, and turbulence equation were treated as convergence when all the residuals reach the criteria. The convergence of residuals for continuity, momentum and energy equation were resolved to levels of 10⁻⁶. The maximum computational iterations are 1000 for meeting convergence criteria and the residuals have a negligible change. Typical structured mesh sizes consisted of 1 x 10⁶ cells with clustering near wall region, which the width of the first cell is 1 x 10⁻⁶m to meet the constrains on the y⁺ values under unit. Comparing mesh size of 1 x 10⁶ and 3 x 10⁶, the area-averaged effectiveness difference between the two mesh sizes was found to vary by only Δη=0.0005 at a level of η=0.35.

**Computational domain:**

**geometry and boundary condition**

The airfoil investigated in this paper is a generic nozzle guide vane of modern highly-loaded high pressure turbine. The two-dimensional cascade geometry was extracted from the airfoil shroud section and stretched in spanwise direction. Table 1 list the key dimensions and the flow conditions of the engine condition. Although previous studies show that the geometry of endwall-vane junction has an important impact on the development of the horseshoe vortex, it’s not involved in this paper and will be discussed in the future investigation. Figure 1 depicts the vane and the slot geometry represented by the black rectangle. This short slot flow from the backward facing step covers only part of the passage. The baseline short slot of height 0.5mm and width 269mm was positioned with its downstream edge 6mm(0.2Cax) upstream of the cascade. When study the effect of axial position, two short slots (AP1, AP2) were positioned at 0.1Cax and 0.3Cax upstream of the vane leading edge respectively.

<table>
<thead>
<tr>
<th>Table1</th>
<th>Vane geometry and flow conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vane chord(C)</td>
<td>0.06m</td>
</tr>
<tr>
<td>Axial chord(Cax)</td>
<td>0.03m</td>
</tr>
<tr>
<td>Pitch/axial chord(P/Cax)</td>
<td>1.8</td>
</tr>
<tr>
<td>Span/axial chord(S/Cax)</td>
<td>1</td>
</tr>
<tr>
<td>Reₘ/ Reₘₐt</td>
<td>1.96 x 10⁷, 8.6 x 10⁴</td>
</tr>
<tr>
<td>Inlet /exit Mach number</td>
<td>0.143, 1.19</td>
</tr>
</tbody>
</table>
All the boundary conditions were set consistent with the design-point flow condition of the turbine nozzle vane. Both the main inlet and slot inlet were assumed fixed mass flow rate to insure the blowing ratio, while the outlet boundary condition was placed 1.5Cax downstream of the trailing edge and assumed constant static pressure. The turbulent energy was set with a turbulence intensity of 5.0%. Periodic boundary conditions were placed along the pitchwise boundaries. No slip and adiabatic conditions were applied on all the wall surface. A symmetry condition was applied at the midspan for reducing the calculating domain.

RESULTS AND DISCUSSION
Aerodynamic analysis
Before discussing the thermal result with slot cooling, some basic aerodynamic result without cooling are first presented in this section to understand the key flow physics relevant to the thermal field. Figure 2 shows the static pressure coefficient distributions along the vane surface at the midspan. The pressure has a negligible decrease until it reaches 0.6Cax and then decreases rapidly to trailing edge on the pressure surface. It is more complicated for suction side that static pressure coefficient changes along vane surface. On the front of the surface there is a sharp pressure decreases then the pressure gets to be unstable at 0.5Cax. After the instability, the pressure rises near the trailing edge. Figure 3 depicts the contour of Mach number where we know that the instability was caused by the impingement of the shock wave emanated from the trailing edge of the neighbouring blade. The shock wave interacts with suction surface boundary layer resulting in the change of static pressure distribution, also leading to a locally considerably increased turning of flow towards the suction side within the passage.

Figure 4 shows the contour of endwall static pressure coefficient. Inclined contour lines, especially near the suction side, reveals the pressure difference between pressure and suction side that drives the cross flow within the passage. Additionally, each contour line has a small bump. The of the
pressure contour lines reflect the trace of the pressure leg of the horseshoe vortex.

The effect of blowing ratio

Figure 5 compares the effect of different blowing ratio on endwall film cooling effectiveness of the short slot. As concluded by previous researchers, with the increase of blowing ratio the integral adiabatic effectiveness increases especially near the vane leading edge and pressure side within passage. Note that due to the existence of a gap between two short slots, the ejected coolant just covers part of the passage area to form an obvious triangle region without coverage. With BR=1.86, leading edge region and pressure side-endwall junction achieves better coolant coverage. It is interesting to find from figure 5 that the change of film cooling effectiveness by increasing blowing ratio from 1.7 to1.86 is much more significant than that from 1 to 1.7, and the increment of coolant mass flux of the late case was several times larger than the former. Figure 6 shows the velocity vectors with the background of static temperature contour at stagnation plane. Figure 6(a) is the result without the short slot injection. The incoming flow boundary layer was separated to form a horseshoe vortex, inducing a small tertiary vortex with opposite sense of rotation. The tertiary vortex is confirmed to have an important effect on the mix process of coolant and main flow. Coolant has a low energy exiting from the short slot and is intensely mixed with main flow when blowing ratio value is 1. Inspections of the figure 6 it can be found the radial mixing height is reduced with the increasing blowing ratio, which gives boundary layer an energy supplement causing the suppression of horseshoe vortex. But the core of the horseshoe moves to further close to leading edge, and the small tertiary vortex is then weakened. The coolant is rolled up by the horseshoe vortex before reaching the vane leading edge, so there leaves a warm ring around the airfoil extends from the stagnation location and along the entire pressure side. When blowing ratio increases to 1.7, the kinetic energy of coolants become stronger but are still rolled up by the horseshoe vortex. Leading edge region near suction side realizes coolant coverage while pressure side-endwall junction is no coolant as a result of the horseshoe vortex as well as the passage vortex that is convected toward the suction side of the vane. The uncovered area near the pressure surface becomes smaller as compared to the case of BR=1.0. Note that for BR=1.86 the contour of adiabatic effectiveness, shown in figure 5 and figure 6, becomes to change dramatically. Coolant jet attached to the endwall and reaching to the leading edge forced the horseshoe vortex to be away from endwall and weaken its strength. After reaching to the leading edge, part of the coolant turns up toward midspan approaches to the vane surface. Thrift and Thole(Thrift et al., 2011) observed this phenomenon that thickening the boundary layer in the near wall region produces a static pressure gradient near the vane-endwall junction that is away from the endwall. A small tertiary vortex with the opposite sense of rotation is also present directly upstream the leading edge.

Figure 7 presents the pitchwise-averaged film cooling effectiveness for different blowing ratio. As indicated in the picture, increasing blowing ratio can increase the film cooling effectiveness, which decreases along axial direction. But it’s
unlikely in experimental condition that the effectiveness keeps decreasing, we have discussed the aerodynamic results that due to the trailing edge shock wave the flow turns to suction side through the passage throat.

Figure 7 Pitchwise-averaged Adiabatic Effectiveness for Different Blowing Ratio

Figure 8 shows the secondary flow vectors and the non-dimensional temperature contours at different axial location within passage. The secondary velocity represents the deviation of the flow relative to that for the midspan inviscid flow. It’s clearly seen that the development of the horseshoe vortex and the passage vortex with and without coolant jets. The core of passage vortex moved closer to pressure side with the increasing of the bowing ratio and can be barely clarified when blowing ratio is over 1.7 at 30%Cax. Passage vortex limits the coolant to spread to pressure side at low blowing ratio and intensified the mixing process. An inspection of the suction side horseshoe vortex, the size becomes smaller with the increasing of the blowing ratio at 30%Cax and swirled in a clockwise direction, but the secondary flow is obviously stronger flowing toward the midspan near suction side at 60%Cax. Meanwhile, the coolant is found to be increases near the suction side due to the pressure difference, which makes the coolant in the passage swept to suction side. The strong velocity vector to suction side close to the endwall indicates the cross flow as well. This upward flow entrained the coolant away from endwall, especially at 90%Cax. So, the development of the secondary flow explains the effect on the distribution of the adiabatic effectiveness. The horseshoe vortex mainly affects the leading edge region while the pressure difference dominates within the passage.
Figure 8 Secondary Flow Vectors and the Thermal Field within Passage at (a) 30%CaX, (b) 60%CaX, (c) 90%CaX

The effect of axial position

The effect of the slot position on cooling performance and stagnation plane flow characteristic are studied by changing the axial position. Three locations of Z/ CaX=0.1(AP1), Z/ CaX=0.2(baseline) and Z/ CaX=0.3(AP3) are studied. Figure 9 and 10 present the adiabatic effectiveness distribution and velocity vector at stagnation plane of different axial position with blowing ratio of 1.7. The phenomenon found in different axial position was similar to that found in different blowing ratio. When the short slot moves closer to the leading edge (AP1), the adiabatic effectiveness gets better on the endwall, especially near the vane leading edge and pressure side. At a location of AP2, the short slot is more far away from the leading edge resulting a longer distance mixed with main flow. Additionally, coolant jet has fewer impact on the horseshoe vortex and is rolled up upstream of the leading edge. But there has little difference of the adiabatic effectiveness distribution between AP2 and baseline axial position. Moving the short slot very close to the leading edge at Z/ CaX=0.1(AP1), the adiabatic effectiveness looks like more uniform and reasonable. Maybe the coolant attaches to the endwall closer so that large area is cooled even downstream of the trailing edge. Figure 10(a) indicates a small tertiary vortex, developed near the leading edge, pushes the horseshoe vortex back over the short slot. As a result, the negative effect of the horseshoe vortex is reduced and the coolant exiting from the short slot is able to impinge the leading edge and climbs up. Consequently, the secondary flow moving towards the endwall due to horseshoe vortex suppress the coolant to prevent separated from it. The coolant cools the endwall rather than mixing with the main flow.

Figure 9 Predicted Adiabatic Effectiveness Levels for Different Axial Position

Figure 10 Flowfield Vectors and Contours of Static Temperature in the Stagnation Plane for Different Axial Position
Figure 11 Pitchwise-averaged Adiabatic Effectiveness for Different Axial Position

Figure 11 presents the pitchwise-averaged adiabatic effectiveness of different axial position. The adiabatic effectiveness of AP1 is significantly better than the other two cases, which are explained in the previous sections. AP2 and baseline case almost have the same adiabatic effectiveness on the entire passage. It can be concluded that the axial position doesn’t change the adiabatic effectiveness obviously when coolant jets are far away from horseshoe vortex.

Figure 12 depicts the result of the effect of the axial position on the secondary flow and the thermal field. The noticeable difference between the two different slot position exists near the pressure side region. At 30%Cax the slot(AP1) nearly located at leading edge has a better effectiveness near pressure side endwall even on the pressure surface. The passage vortex entrained the coolant on pressure surface onto the endwall surface which can better protect endwall at the rear of the passage. The coolant from AP1 remains better attached to the endwall than that from AP2 as a result of the reducing strength of the passage vortex. But it looks have the same strength downstream of the passage. Contrary to the passage vortex, the horseshoe vortex of suction side is stronger when reduce the axial distance of slot and leading edge at 30%Cax. At 60%Cax and 90%Cax, the pattern with changing the axial position is much similar with the changing the blowing ratio. It also concludes that the pressure difference dominates the distribution of the secondary flow therefore the adiabatic effectiveness within the passage.

The effect of coolant incidence angle

The effect of incidence angle of main flow on adiabatic effectiveness has been investigated by several researchers, while the author hasn’t found the research about the effect of coolant incidence angle. Figure 15 shows the adiabatic effectiveness contours of different coolant incidence angle. The cooling patterns are similar with the incidence angle changing from $a=-10\text{deg}$ to $a=10\text{deg}$. Figure 15(c) shows a better coverage near the vane leading edge and pressure side in the front of the passage. Unlike aforementioned effect of axial position of the short slot, pressure side fails to be cooled in the rear part of the passage due to the reduced axial velocity, which is more distorted by the cross flow. The horseshoe vortex dominates the adiabatic effectiveness near the leading edge region while the cross flow and passage vortex dominates the adiabatic effectiveness within the passage.
Figure 16 gives the pitchwise-averaged adiabatic effectiveness result. Result shows little change when changing the coolant incidence angle.

CONCLUSIONS

Numerical studies were carried out to investigate the film cooling performance of short slot. The numerical model simulates a nozzle guide vane of modern highly-loaded HP-turbine in an engine-scale two-dimensional cascade set under a transonic aerodynamic design condition. Results for three different slot blowing ratio are presented at a nominal upstream location. In addition, results are presents for further upstream and further downstream from the nominal location. Several coolant incidence angles are considered while keeping blowing ratio constant. Calculated results show that the short slot has an important effect on the formation and development of the horseshoe vortex. The main deficiency of the short slot is a triangle shaped region in the front of the passage without coolant coverage.

At a nominal upstream location, the case at BR=1.86 provides a substantial improvement in the endwall adiabatic effectiveness over BR=1.7, especially near the vane leading edge and pressure side-endwall junction. Pitchwise-averaged adiabatic effectiveness shows the huge difference even with few coolants increasing. Moving the short slot at Z/Cax=0.1 upstream of the leading edge, the region of low static temperature along spanwise direction shows coolants are reduced to mixing with main flow. The stagnation plane flowfields indicate a small tertiary vortex is formed close to leading edge, which is found have an important effect on the mixing process. The small tertiary vortex pushes the horseshoe vortex more upstream of the leading edge. Changing the coolant incidence angle show little effect on the cooling effectiveness in this paper.

NOMENCLATURE

\( a \) \hspace{1cm} coolant incidence angle
\( A P1, AP2 \) \hspace{1cm} different axial position
\( BR \) \hspace{1cm} blowing ratio
\( C \) \hspace{1cm} true vane chord
\( Cax \) \hspace{1cm} axial vane chord
\( CP1, CP2 \) \hspace{1cm} different circumferential position
\( C_p \) \hspace{1cm} static pressure coefficient, \( (p-p_m)/(\frac{1}{2} \rho_{in} U^2) \)
\( Ma \) \hspace{1cm} mach number
\( P \) \hspace{1cm} vane pitch
\( p \) \hspace{1cm} static pressure
\( p_0 \) \hspace{1cm} total pressure
\( Re \) \hspace{1cm} Reynolds number, \( \rho U_s C_{in} / \mu \)
\( S \) \hspace{1cm} vane span height
\( T \) \hspace{1cm} static temperature
\( U \) \hspace{1cm} streamwise velocity
\( X \) \hspace{1cm} pitchwise direction
\( Z \) \hspace{1cm} axial direction
\( \eta \) \hspace{1cm} adiabatic effectiveness
\( \mu \) \hspace{1cm} viscosity
\( \rho \) \hspace{1cm} density

Subscripts

\( in \) \hspace{1cm} mainflow inlet
\( exit \) \hspace{1cm} mainflow outlet
\( \infty \) \hspace{1cm} freestream velocity at entrance

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